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DESIGN OF REGENERATORS FOR HEAT TRANSFER FLUIDS WITH VARIABLE THERMOPHYSICAL PROPERTIES

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Theoretical formulas are obtained for determining the heat transfer surface of a regenerator in the case of the heat capacity of one of the heat transfer fluids, and also the heat transfer coefficient varying substantially through the equipment. The effectiveness of the results is illustrated by concrete examples.

In many cases of heat transfer between fluids taking place in regenerators, the physical properties of the heat transfer fluids vary so much that appreciable errors result from inserting into the theoretical formulas values averaged over the temperature range. In the most general case, not only the heat transfer coefficient, but also the heat capacity of the fluid are functions of temperature. The Altenkirch [1] method is then recommended, according to which

$$F/G_2 = \int_{t''_i}^{t''_f} c_p dt''/k(t''-t'). \tag{1}$$

Calculations based on (1) may be performed by either numerical or graphical integration, a laborious operation involving evaluation of the heat transfer coefficient for various temperatures, as well as determination from tables of appropriate values of the specific heat. It should be noted that in a number of cases (close to the critical state) these values should be determined using i-T diagrams.

In this situation, it is natural that a number of investigators should have attempted to develop analytical methods that would permit allowance to be made for changes of the physical properties of the heat transfer fluids during heat transfer. One of the earliest methods is that proposed by Shak [2], based on an assumed power law of the type

$$k = a + bF_x^n, \tag{2}$$

it being recommended that the exponenet is determined experimentally. This considerably complicates the application of relation (2) to equipment design. In the special case n = 1 the calculation reduces to introducing an arithmetic mean value of k for the ends of the apparatus.

The method proposed in [3] is based on the assumption of a linear dependence of the heat transfer coefficient on the temperature of one of the heat transfer fluids

$$k = a + bt''. \tag{3}$$

This assumption, along with the assumption that the specific heat is constant, leads to the relation

$$\frac{Q}{F} = [k_i(\Delta t)_f - k_f(\Delta t)_i]\left[\ln\frac{k_i(\Delta t)_f}{k_f(\Delta t)_i}\right]^{-1}, \tag{4}$$

which agrees much better with the exact formula (1) than does the usual Grashof formula.

In cases when the heat transfer coefficient is practically constant, but there is a substantial variation of specific heat, apart from (1), graphical methods are recommended, based on the use of i-T diagrams [4, 5] for those substances for which such diagrams exist. An analytical examination of the variation of specific heat was made in [6], but, because of an incorrect assumption,* the design formulas obtained require revision. However, as noted above, in practice we may encounter a simultaneous appreciable variation of specific heat and heat transfer coefficient. The following discussion is devoted to an analytical examination of this case. We shall restrict our consideration to steady parallel flow of two heat transfer fluids, heat transfer between which is described by the system of equations [6]

$$\begin{gathered} t' = t'' \pm \frac{W_2}{ks}\,\frac{dt''}{dx}, \\ t'' = t' + \frac{W_1}{ks}\,\frac{dt'}{dx}, \end{gathered} \tag{5}$$

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*This error was kindly brought to the author's attention by K. D. Voskresenskii.

where the minus sign refers to the counterflow case.

In these equations k is a variable and may be assumed, as will be shown below, to be a function of the temperature of one of the heat transfer fluids, e.g., t". The water equivalent W_2 of one of the fluids is also a function of t".

In general, variation of the specific heat of not one, but both fluids is possible. In practice, however, it is met rather seldom and will not be examined here, although to obtain a solution for this situation is not difficult in principle.

Going to a dimensionless representation, using the notation

$$k/k_0 = \varphi(t''), \quad c_2/c_{20} = W_2/W_{20} = \psi(t'') \tag{6}$$

and replacing system (5) by one second-order equation, we obtain

$$\frac{d^2t''}{dv_x^2} + \left[\varphi \pm R_{12}\frac{\varphi}{\psi} + \frac{d(\ln\psi/\varphi)}{dt''}\,\frac{dt''}{dv_x}\right]\frac{dt''}{dv_x} = 0. \tag{7}$$

In (6), k_0 and c_{20} are fixed values of these factors at one end of the regenerator. If we assume that k_0 and c_{20} relate to the section where heat transfer fluid II enters the equipment, then $\varphi(t''_i) = 1$; $\Psi(t''_i) = 1$.

The boundary conditions for (7) will be:
for parallel flow

$$t''|_{v_x=0} = t''_i;\ \left.\frac{dt''}{dv_x}\right|_{v_x=0} = R_{12}(t'_i - t''_i); \tag{8}$$

for counterflow

$$t''|_{v_x=v} = t''_i;\ \left.\frac{dt''}{dv_x}\right|_{v_x=v} = -R_{12}(t'_f - t''_i), \tag{9}$$

assuming that fluid II flows in the negative x direction. Following the substitution $\frac{dt''}{dv_x} = p$, Eq. (7) becomes

$$\frac{dp}{dt''} + \frac{d(\ln\psi/\varphi)}{dt''}\,p = -\varphi\left(1 \pm \frac{R_{12}}{\psi}\right), \tag{10}$$

and, in accordance with (8) and (9), its solution

$$p \equiv \frac{dt''}{dv_x} = \frac{\varphi}{\psi}\left[-\int(\psi \pm R_{12})\,dt'' + C\right], \tag{11}$$

must satisfy the boundary condition

$$p|_{t''=t''_i} = R_{12}(t'_i - t''_i) \tag{12}$$

for parallel flow and

$$p|_{t''=t''_i} = -R_{12}(t'_f - t''_i) \tag{13}$$

for counterflow.

From (11), using (12) and (13), we may determine, for the two cases of direction of flow of the heat transfer fluids

$$v \doteq \int_{t''_i}^{t''_\kappa} \frac{(\psi/\varphi)\,dt''}{\int_{t''_i}^{t''} \psi\,dt'' + R_{12}(t''_i - t'') - R_{12}(t''_i - t'_i)} \quad \text{for parallel flow,}$$

$$v = \int_{t''_f}^{t''_i} \frac{(\psi/\varphi)\,dt''}{R_{12}(t''_i - t'_f) - R_{12}(t''_i - t'') - \int_{t''_i}^{t''} \psi\,dt''} \quad \text{for counterflow,} \tag{14}$$

if the form of functions $\Psi(t'')$ and $\varphi(t'')$ is known.

The ratio of the heat transfer coefficients will be

$$\varphi \equiv \frac{k}{k_0} = \left(1 + \frac{\delta}{\lambda}\alpha_{20} + \frac{\alpha_{20}}{\alpha_{10}}\right) \Bigg/ \left(\frac{\alpha_{20}}{\alpha_2} + \frac{\delta}{\lambda}\alpha_{20} + \frac{\alpha_{10}}{\alpha_1}\,\frac{\alpha_{20}}{\alpha_{10}}\right).$$

The heat transfer coefficient for turbulent flow in tubes may be written in the form

$$\alpha = A\,\lambda^{0.6}(c_p/\mu)^{0,4} \tag{15}$$

and therefore

$$\frac{\alpha_0}{\alpha} = \left(\frac{\lambda_0}{\lambda}\right)^{0.6}\left(\frac{\mu}{\mu}\right)^{0.4}\left(\frac{c_{p0}}{c_p}\right)^{0.4}. \tag{16}$$

If the heat transfer fluid is a gas, the ratios of thermal conductivities and viscosities may be replaced by their appropriate temperature dependences [7, 8]. With respect to the dependence of specific heat on temperature, the form usually given is a polynomial. For the problem discussed here, for convenience of later calculation, we put the specific heat dependence in the same form as that assumed for viscosity and thermal conductivity, i. e.,

$$c_p/c_{p0} = (T/T_0)^p. \tag{17}$$

This relation was checked against data presented in [10], and it was found that over quite a wide range of temperature it gives errors not greater than a few percent (table). In view of the fact that in (16) the ratio of specific heats has the exponent 0.4, this error is quite acceptable.

Values of the Exponent p is (17) for Some Gases and Vapors at a Pressure of $10^5\,\mathrm{N/m^2}$

Gas	p	Temperature range	Error %
Methane	0.59	0—500° C	6
Sulfur anhydride	0.18	0—800° C	4
Ammonia	0.31	300—600° K	3
Propane	0.92	250—600° K	4
Ethyl alcohol	0.73	100—600° C	1
Air	0.19	500—2000° K	7

Thus, instead of (16), we obtain

$$\frac{\alpha_0}{\alpha} = \left(\frac{T}{T_0}\right)^{0.4m-0.4p-0.6n} = $$

$$= \left(1 + \frac{T - T_0}{T_0}\right)^{0.4m-0.4p-0.6n}$$

where m and n are the exponents in the temperature relations for the viscosity and thermal conductivity of the gases.

If for design purposes, the regenerator is divided into sections, in each of which the ratio $(T - T_0)/T_0$ is sufficiently small, then the last expression may be linearized, i. e.,

$$\frac{\alpha_0}{\alpha} \approx 1 + \frac{0.4m - 0.4p - 0.6n}{T_0}(T - T_0) = 1 + b_g(T - T_0).$$

It will be shown below that the number of such sections is not more than two, even in cases of strong variation of the heat transfer coefficients and specific heats along the length of the equipment.

If the heat transfer fluid is a liquid, the variation of the right side of (16) will mainly be determined by the temperature dependence of the viscosity, since the product $\lambda^{0.6}c_p^{0.4}$ changes little with temperature. For example, for water in the range 200-350°C, the product $\lambda^{0.6}c_p^{0.4}$ changes by 7%, for ethyl alcohol in the range 78.4-231.5° C – by 1%, for diphenyl mixture – by 1% (100-300°), and for tetracresyloxysilane – by 15% with temperature change from 100 to 300°C.

According to the Andrade theory [9], the viscosity of a liquid is related to its temperature and density by

$$\mu = A\rho^{1/3}\exp(B\rho/T).$$

Taking account of the small dependence of density ρ on T, and replacing it by its mean value ρ_m, we obtain, in place of (16)

$$\frac{\alpha_0}{\alpha} = \exp\left[\frac{0.4B\rho_m}{T_0 T}(T - T_0)\right].$$

Restricting our attention, as with gaseous heat transfer fluids, to a section of the equipment in which the exponent is sufficiently small, we obtain the following approximate relation:

$$\frac{\alpha_0}{\alpha} \approx 1 + \frac{0.4 B \rho_{\mathrm{m}}}{T_0 T}(T - T_0) = 1 + b_l (T - T_0). \tag{18}$$

The approximate nature of this result is due, apart from the basic assumptions, to the fact that b_l was written in (18) as a constant, although it depends on temperature; it is therefore desirable to give it the value $b_{l \cdot \mathrm{m}}$, corresponding to $(T + T_0)/2$.

Thus, the dependence of α_0/α may be represented in the first approximation as a linear function of the difference between the temperature of the heat transfer fluid at the inlet to the equipment and the variable temperature.

Using the equilibrium equation $t' - t''_0 = R_{21}(t''_0 - t'')$, which will be sufficiently accurate on a limited section of the heat transfer surface even in the case of variable specific heat, we obtain:

$$\frac{k}{k_0} = \frac{1}{1 - b(t''_0 - t'')}, \quad b = \frac{b_2 - (\alpha_{20}/\alpha_{10})\, b_1 R_{21}}{1 + \delta\alpha_{20}/\lambda + \alpha_{20}/\alpha_{10}}. \tag{19}$$

Comparing (19) and (3), we see that for small values of $b(t''_0 - t'')$, both expressions will give practically identical results. Relation (19), however, more correctly reflects the physical picture of heat transfer when the thermophysical properties of both media vary.

The second equation of (19) allows us to obtain the condition for a constant heat transfer coefficient. This corresponds to $b = 0$, i. e., $b_2/b_1 = \alpha_{20}R_{21}/\alpha_{10}$. In examining the temperature dependence of the ratio of water equivalents or specific heats c_2/c_{20}, we assumed a linear relation of the form

$$c_2/c_{20} = 1 - a(t''_0 - t''), \tag{20}$$

which is a first approximation for the quadratic polynomial in temperature that describes the variation of the specific heats of liquids and gases.

Expressions (19) and (23) are a form of the functions Ψ and φ defined by (6) and entering into (14).

Substituting (19) and (20) into (14) and integrating, we obtain a formula for calculating the heat transfer surface that takes into account the simultaneous variation of the specific heat and heat transfer coefficient with temperature:

$$\begin{aligned} v \equiv \frac{k_{\mathrm{i}}F}{W_1} = {} & \pm 2\left(1 - \frac{k_{\mathrm{i}}}{k_{\mathrm{f}}}\right) \mp \left[1 - (1 \pm 2R_{12})\frac{1 - k_{\mathrm{i}}/k_{\mathrm{f}}}{1 - c_{2\mathrm{f}}/c_{2\mathrm{i}}}\right] \ln\left\{\frac{1}{\beta}\left[\beta - \right.\right. \\ & \left.\left. - 1 \mp \frac{1}{2} R_{21}\left(1 + \frac{c_{2\mathrm{f}}}{c_{2\mathrm{i}}}\right)\right]\right\} \pm \frac{1}{\varkappa}\left\{(1 \pm R_{12})(1 \pm 2R_{12})\frac{1 - k_{\mathrm{i}}/k_{\mathrm{f}}}{1 - c_{2\mathrm{f}}/c_{2\mathrm{i}}} \mp \right. \\ & \left. \mp R_{12}\left[1 + 2\beta\left(1 - \frac{k_{\mathrm{i}}}{k_{\mathrm{f}}}\right)\right]\right\} \ln \frac{1 \pm R_{21}(1 - \varkappa) - 2\beta}{1 \pm R_{21}(1 + \varkappa) - 2\beta}, \end{aligned} \tag{21}$$

where for parallel flow the upper sign should be taken and

$$\beta = \frac{t''_{\mathrm{i}} - t'_{\mathrm{i}}}{t''_{\mathrm{i}} - t''_{\mathrm{f}}}, \quad \varkappa = \sqrt{(R_{12} + 1)^2 - 2R_{12}\beta(1 - c_{2\mathrm{f}}/c_{2\mathrm{i}})}, \tag{22}$$

and for counterflow – the lower sign and

$$\beta = \frac{t''_{\mathrm{i}} - t'_{\mathrm{f}}}{t''_{\mathrm{i}} - t''_{\mathrm{f}}}, \quad \varkappa = \sqrt{(R_{12} - 1)^2 + 2R_{12}\beta(1 - c_{2\mathrm{f}}/c_{2\mathrm{i}})}. \tag{22a}$$

In substituting numerical values in (21), it should be noted that the quantities k_{i} and $c_{2\mathrm{i}}$ relate to the section where the heat transfer fluid with variable specific heat enters the equipment.

Formula (21) is not valid for all cases of heat transfer, as can be seen from an examination of the integrals in (14), after substituting values of φ and Ψ from (19) and (20). If $c_{2\mathrm{f}}/c_{2\mathrm{i}} \geq 1$ in parallel flow or $c_{2\mathrm{f}}/c_{2\mathrm{i}} \leq 1$ in counterflow, Eq. (21) is applicable. These conditions are very rigorous. In addition, Eq. (21) is also applicable when the inequality

$$c_{2\mathrm{f}}/c_{2\mathrm{i}} \leqslant 1 + (1 \pm R_{12})^2/2R_{12}\beta, \tag{23}$$

is fulfilled, which occurs when $R_{12} \to \infty$ ($R_{21} = 0$), i. e., when the temperature of the heat transfer fluid I does not vary along the heat transfer surface.

However, if

$$c_{2\mathrm{f}}/c_{2\mathrm{i}} \geqslant 1 + (1 \pm R_{12})^2/2R_{12}\beta, \tag{23'}$$

then Eq. (21) is unsuitable, and in this event from (14) we obtain

$$\begin{gathered}\frac{k_\mathrm{i}F}{W_1} = \pm 2\left(1 - \frac{k_\mathrm{i}}{k_\mathrm{f}}\right) \mp \\ \mp \left[1 - (1 \pm 2R_{12})\frac{1 - k_\mathrm{i}/k_\mathrm{f}}{1 - c_{2\mathrm{f}}/c_{2\mathrm{i}}}\right] \ln\left\{\frac{1}{\beta}\left[\beta - \right.\right. \\ \left.\left. - 1 \mp \frac{1}{2}R_{21}\left(1 + \frac{c_{2\mathrm{f}}}{c_{2\mathrm{i}}}\right)\right]\right\} \pm \\ \pm \frac{2}{\varkappa_1}\left\{(1 \pm R_{12})(1 \pm 2R_{12})\frac{1 - k_\mathrm{i}/k_\mathrm{f}}{1 - c_{2\mathrm{f}}/c_{2\mathrm{i}}} \mp \right. \\ \left. \mp R_{12}\left[1 + 2\beta\left(1 - \frac{k_\mathrm{i}}{k_\mathrm{f}}\right)\right]\right\} \operatorname{arc\,tg}\frac{\varkappa_1}{1 \mp R_{12}(2\beta - 1)},\end{gathered} \tag{24}$$

where

$$\varkappa_1 = \sqrt{\pm 2R_{12}\beta(c_{2\mathrm{f}}/c_{2\mathrm{i}} - 1) - (1 \pm R_{12})^2}.$$

Let us examine some special cases.

The temperature of the heat transfer fluid with constant specific heat does not vary along the heat transfer surface ($R_{21} = 0$). In this case instead of (21) and (24) we obtain

$$\begin{gathered}u \equiv k_\mathrm{i}F/W_{2\mathrm{i}} = (1 - k_\mathrm{i}/k_\mathrm{f}) + (1 - c_{2\mathrm{f}}/c_{2\mathrm{i}}) + \frac{1}{2}(1 - k_\mathrm{i}/k_\mathrm{f})(1 - \\ - c_{2\mathrm{f}}/c_{2\mathrm{i}})(1 - 2\beta) + [1 - \beta(1 - k_\mathrm{i}/k_\text{к})][1 - \beta(1 - \\ - c_{2\mathrm{f}}/c_{2\mathrm{i}})] \ln[(\Delta t)_\mathrm{i}/(\Delta t_\mathrm{f})].\end{gathered} \tag{25}$$

If k = const, then

$$u = 1 - c_{2\mathrm{f}}/c_{2\mathrm{i}} + [1 - \beta(1 - c_{2\mathrm{f}}/c_{2\mathrm{i}})] \ln[(\Delta t)_\mathrm{i}/(\Delta t)_\mathrm{f}]. \tag{25'}$$

If c_p = const, then

$$u = 1 - k_\mathrm{i}/k_\mathrm{f} + [1 - \beta(1 - k_\mathrm{i}/k_\mathrm{f})] \ln[(\Delta t)_\mathrm{i}/(\Delta t)_\mathrm{f}]. \tag{25''}$$

The heat transfer coefficient varies little along the heat transfer surface (k ≈ const). In this case instead of (21) we obtain

$$\begin{gathered}v = \mp \ln\left\{\frac{1}{\beta}\left[\beta - 1 \mp \frac{1}{2}R_{21}(1 + c_{2\mathrm{f}}/c_{2\mathrm{i}})\right]\right\} - \\ - \frac{R_{12}}{\varkappa} \ln\frac{1 - 2\beta \pm R_{12}(1 - \varkappa)}{1 - 2\beta \pm R_{12}(1 + \varkappa)},\end{gathered} \tag{26}$$

and instead of (24)

$$\begin{gathered}v = \mp \ln\left\{\frac{1}{\beta}[\beta - 1 \mp R_{21}(1 + c_{2\mathrm{f}}/c_{2\mathrm{i}})]\right\} - \\ - \frac{2R_{12}}{\varkappa_1} \operatorname{arctg}\frac{\varkappa_1}{1 \mp R_{12}(2\beta - 1)},\end{gathered} \tag{27}$$

where $c_{2\mathrm{i}}$ corresponds to the value of the specific heat at the inlet to the apparatus.

The specific heat varies little along the heat transfer surface (c_p ≈ const). When $c_{2\mathrm{f}} = c_{2\mathrm{i}}$, (21) and (24) give an indeterminate form of the type $\infty - \infty$, on expanding which we find

$$v = \pm \frac{1}{1 \pm R_{12}} \left\{ 1 - \frac{k_i}{k_f} - \left[1 \mp \frac{\beta R_{12}}{1 \pm R_{12}} \times \right. \right.$$

$$\left. \left. \times \left(1 - \frac{k_i}{k_f} \right) \right] \ln \left(1 - \frac{1 \pm R_{21}}{\beta} \right) \right\},$$

where k_i corresponds to admission of a fluid with initial temperature t_i'', which should be borne in mind when calculating the parameter β.

When $R_{12} = 1$ for counterflow, by expanding the indeterminate form in (28), we obtain

$$v = \frac{1}{2\beta} \left(1 + \frac{k_i}{k_f} \right). \tag{29}$$

Calculations show that in using (21), (24) etc. there is, in most cases, no need to divide the heat transfer surface into sections, and with sufficient accuracy we may confine our attention to data on the values of the heat transfer coefficients and specific heats at the ends of the equipment. In important calculations it may be justifiable to divide the surface into two parts, having calculated the heat transfer coefficient for an intermediate section of the equipment; if the result then differs only slightly from that obtained by calculating the heat exchanger as a whole, further subdivision of the heat transfer surface will not be necessary.

Let us illustrate the effectiveness of the proposed method by two examples.

I. The high-temperature heat transfer fluid tetracresyloxysilane is used for evaporating a liquid at $t' = 100°C$.

The temperature of the fluid at the outlet is $t_f'' = 105°C$, and the temperature at the inlet $t_i'' = 300°C$, is such as to avoid decomposition [11]. The enthalpies, heat flux, and specific heats corresponding to these temperatures are, respectively, $i_i'' = 9.11 \cdot 10^5$ joule/kg, $i_f'' = 2.21 \cdot 10^5$ joule/kg, $Q = G_2(9.11 - 2.21) \cdot 10^5 = 6.90 \cdot 10^5\ G_2$ watt, and $c_{2i} = 4.65 \cdot 10^3$ joule/kg·deg, $c_{2f} = 2.52 \cdot 10^3$ joule/kg·deg. Since the thermal resistance of the boiling liquid and the wall may be neglected, the heat transfer coefficient will be given by (15).

Let us assume that the quantity A, which depends on the mass flow rate of the fluid and the tube diameter, has been determined and that $k_i = 1000$ W/m^2 · deg; then $k_f = 407$ W/m^2 · deg, which follows from (15) if we assume the thermophysical properties are evaluated at 105° C.

Thus, in the example given we have a change in the specific heat and heat transfer coefficient by a factor of two or more. Since the temperature of the boiling liquid is constant ($R_{21} = 0$), formula (25) should be used, from which we obtain, after substituting numerical values, $k_i F/W_{2i} = 4.21$ whence $F = 19.5\ G_2$ m^2.

The exact solution obtained by graphical integration of (1) gives $k_i F/W_{2i} = 4.47$, i.e., a value 6% higher, which is not significant, since this difference falls within the limits of accuracy of the determination of the heat transfer coefficient.

If (4) is used, we obtain $F = 25.2\ G_2$, i.e., a value 22% greater than that given by the exact solution. This difference would be even larger if the specific heat remained unchanged.

II. This example is borrowed from [4]. In a counterflow apparatus there is heat exchange between air at $G_2 = 28$ m/sec and $p = 80.10^5$ N/m^2 and nitrogen at $p = 1.2 \cdot 10^5$ N/m^2.

The air temperatures are $t_i'' = 303°K$; $t_f'' = 155°K$, and those for nitrogen $t_i' = 80.3°K$; $t'_f = 298°K$. We find the specific heat of air at the ends of the apparatus by numerical differentiation with respect to the i-T diagram: $c_{2i} = 1.12 \cdot 10^3$ J/kg · deg, $c_{2f} = 3.42 \cdot 10^3$ J/kg · deg.

We find $W_{2i} = 3190$ W/deg. Corresponding to the given thermal flux $Q = 6300$ W we find $W_1 = Q/\Delta t' = 6300/218 = 28.9$ W/deg and $R_{21} = 3190/28.9 = 110.5$. From (22) we determine $\beta = 0.0338$. We verify that (23) is satisfied. From (26), neglecting the second term because in this case, it is small, we obtain $v = 8.77$, whence $kF = 28.9 \times 8.77 = 254$ W/deg. The accurate, but laborious method described in [4] gives the result $kF = 280$ W/deg, i.e., the discrepancy is 10%, although the change in specific heat is very considerable ($c_{2f}/c_{2i} = 3.06$).

These examples indicate that in the range of variation of k_f/k_i and c_{2f}/c_{2i} encountered in practice, use of the recommended formulas should give quite acceptable results without recourse to zonal heat transfer calculations; this becomes expedient when k_f/k_i and c_{2f}/c_{2i} vary by more than a factor of three, but even then it is sufficient to divide the equipment into only two zones.

NOTATION

F — heat exchange surface; W — water equivalent of heat transfer fluid; Δt — temperature difference of heat transfer fluids; k — heat transfer coefficient; s — perimeter of surface; $R_{12} = 1/R_{21} = W_1/W_2$; $v_x = k_i F_x/W_1$; v — value of v_x corresponding to total heat transfer surface. Subscripts: 1 and 2, and ' and " refer to the first and second heat transfer fluids, i and f to the sections where the fluids enter and leave the equipment, 0 to one of the ends of the equipment.

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